



Performance characteristic of a Stirling refrigeration cycle in micro/nano scale

Wenjie Nie^{a,*}, Jizhou He^b, Jianqiang Du^a

^a School of Computer, Jiangxi University of Traditional Chinese Medicine, Nanchang, 330006, People's Republic of China

^b Department of Physics, Nanchang University, Nanchang, 330047, People's Republic of China

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ABSTRACT

The aim of the paper is to present the performance characteristics of a Stirling refrigeration cycle in micro/nano scale, in which the working substance of cycle is an ideal Maxwellian gas. Due to the quantum boundary effect on the gas particles confined in the finite domain, the cycle no longer possesses the condition of perfect regeneration. The inherent regenerative losses, the refrigeration heat and coefficient of performance (COP) of the cycle are derived. It is found that, for the micro/nano scaled Stirling refrigeration cycle devices, the refrigeration heat and COP of cycle all depend on the surface area of the system (boundary of cycle) besides the temperature of the heat reservoirs, the volume of system and other parameters, while for the macro scaled refrigeration cycle devices, the refrigeration heat and COP of cycle are independent of the surface area of the system. Variations of the refrigeration heat ratio r_R and the COP ratio r_ϵ with the temperature ratio τ and volume ratio r_V for the different surface area ratio r_A are examined, which reveals the influence of the boundary of cycle on the performance of a micro/nano scaled Stirling refrigeration cycle. The results are useful for designing of a micro/nano scaled Stirling cycle device and may conduce to confirming experimentally the quantum boundary effect in the micro/nano scaled devices.

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1. Introduction

In classical thermodynamics, the cycle models of various heat engines or refrigerators using gases as working substance are based on the classical ideal gas equation of state and the classical law for heat transfer between the reservoirs and the working substance. For a number of real cycle devices, the predictions of these models about the performance of cycles are good enough since in general on the one hand, the temperature inside the working substance is high and its density is low enough, and on the other hand, the size of the considered system is large enough. However, at some special conditions i.e. sufficiently low temperature or high density conditions, quantum degeneracy of gas particles becomes important and causes differentiation of behaviors of the gas cycles since the states of gases deviate from the classical ideal gas equation of state. In particular, the system, called quantum gas system, should obey the laws of quantum statistical mechanics instead of the classical one. By considering the quantum degeneracy of ideal gases, the performance characteristics of the power or refrigeration cycles working with ideal quantum gases have been analyzed [1–6]. Moreover, the finite-time thermodynamic theories have been used for analyzing the performance of the quantum heat engines [7–12].

Most of the studies about the performance of the gas cycle always neglect the influence of boundary of cycle. That is, the performance of the cycle is independent of the boundary of cycle. For a macro scaled gas cycle system, the results are valid. However, recent advances in nanotechnology [13–15] make the thermodynamic behavior of gases in finite

* Corresponding author.

E-mail address: henameiswen@sina.com (W. Nie).

scale a current issue [16–21]. In the literature, it has been found that thermal wavelength of gas atoms confined in a micro/nano scaled domain may be comparable with the size of the system and the quantum boundary effect of gas particles becomes important [16–18]. Especially, based on the boundary effect, a micro/nano thermosize gas cycle, which is the analog of thermoelectric generator based on the thermoelectric effect, is designed theoretically and may conduce to confirming experimentally the existence of the boundary effect [22,23]. Thus, even for the classical gas cycle systems in finite scale, similar to the influence of quantum degeneracy on the classical cycle, quantum boundary effect of gas particles in a finite domain cannot be neglected and may cause differentiation of behaviors of the gas cycles. And the difference may be measured experimentally by designing a gas cycles i.e. the Stirling refrigeration cycle devices in micro/nano scale. Then, it is necessary to analyze further the performance characteristics of the classical Stirling refrigeration cycle in micro/nano scale, in which the quantum boundary effect has not been previously considered. In the present paper, a micro/nano scaled Stirling refrigeration cycle working with ideal Maxwellian gases is considered. The inherent regenerative losses, refrigeration heat and coefficient of performance of the cycle are derived using the thermodynamic relations of the finite domain. It is seen that they are different from those for the general Stirling cycle, which works at a macro scaled system.

The main purpose of the paper is to present the performance characteristics of a micro/nano scaled Stirling refrigeration cycle and analyze the influence of the boundary i.e. the surface area of cycle on the performance of the cycle by some special examples. The results attained here provide some guidance for designing of a micro/nano scaled heat exchange device and may also conduce to confirming experimentally the quantum boundary effect in the micro/nano scaled devices.

2. Thermodynamic properties of an ideal gas confined in a finite domain

Based on Maxwell–Boltzmann statistics, the free energy of an ideal Maxwellian gas confined in a finite domain which shape can be rectangle, cylinder or sphere is given [16–18]

$$F = -Nk_bT \left[\ln \left(\frac{V\pi^{3/2}}{8NL_c^3(T)} \right) + 1 \right] + \frac{A}{V} \frac{Nk_bTL_c(T)}{2\sqrt{\pi}}, \quad (1)$$

where $L_c(T) = h/\sqrt{8mk_bT}$ is one half of the most probable de Broglie's wave length of the particles at temperature T , h is Planck's constant, m is the atomic mass and k_b is Boltzmann's constant, N is the total number of particles in the whole confinement volume, A is the surface area of the domain and V is the volume of the domain. The first term in Eq. (1) is the classical bulk free energy of gas confined in the finite domain, while the second term is the quantum surface energy, which unavoidably appears and depends strongly on the boundary i.e. surface area of the domain A when the size of the domain is comparable with the mean de Broglie's wavelength of the particles. Further, these additional control variables on the state functions may make the ideal gas confined in a finite domain deviate from its classical behavior i.e. the gas confined in a finite domain cannot fill all the space of the domain and the density goes to zero within the layer approached the boundary of domain [18]. It is interesting that the nature of gases confined in a finite domain will render a realistic thing to design a mechanical gas cycle devices without any physical contact between the piston and cylinder of the cycle. Further, the mechanical friction between the piston and cylinder can be decreased largely. In addition, because of the boundary effect on the thermodynamic state functions of a gas confined in a finite domain, the performance characteristic of the cycle i.e. a Stirling refrigeration cycle in micro/nano scale is different from the macro scaled Stirling refrigeration cycle, such as, the changing of the refrigeration heat and the COP of cycle. That is, it may be observed that the quantum boundary effect of gas particles in a finite domain will cause the differentiation of behaviors of the classical gas cycles.

The internal energy and entropy of an ideal gas confined in a finite domain are, respectively, given by

$$E = \frac{3}{2}Nk_bT + \frac{A}{V} \frac{Nk_bTL_c(T)}{4\sqrt{\pi}} \quad (2)$$

and

$$S = Nk_b \left[\ln \left(C \frac{VT^{3/2}}{N} \right) + \frac{5}{2} \right] - \frac{A}{V} \frac{Nk_bL_c(T)}{4\sqrt{\pi}}, \quad (3)$$

where $C = (2\pi mk_b)^{3/2}/h^3$. From Eq. (2), one obtains the heat capacity at constant volume as

$$C_V = \left(\frac{\partial E}{\partial T} \right)_V = \frac{3}{2}Nk_b + \frac{A}{V} \frac{Nk_bL_c(T)}{8\sqrt{\pi}}. \quad (4)$$

Using Eqs. (3) and (4), one can calculate the amounts of heat exchange in an isothermal and isochoric process. They are given by

$$Q_{ab} = T(S_b - S_a) = Nk_bT \left[\ln \frac{V_b}{V_a} + \left(\frac{A_a}{V_a} - \frac{A_b}{V_b} \right) \frac{L_c(T)}{4\sqrt{\pi}} \right] \quad (5)$$

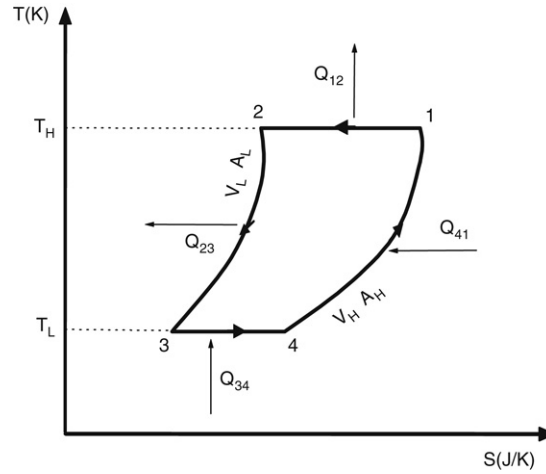


Fig. 1. Temperature–entropy diagram of a micro/nano scaled Stirling refrigeration cycle.

and

$$Q_{ab}^V = \int_a^b C_V(T, A, V) dT = \frac{3}{2} Nk_b (T_b - T_a) + \frac{Nk_b}{8C^{1/3}} \frac{A}{V} (\sqrt{T_b} - \sqrt{T_a}), \quad (6)$$

where the subscripts a and b refer to the initial and final states, respectively. With the help of the thermodynamic relations in the finite domain, we can calculate the coefficient of performance (COP) of a micro/nano scaled Stirling refrigeration cycle working with ideal gases and discuss the influences of boundary on the inherent regenerative losses in two isochoric processes, the refrigeration heat and the COP of the cycle.

3. A micro/nano scaled Stirling refrigeration cycle

A micro/nano scaled Stirling cycle is composed of two isothermal and two isochoric processes. The temperature–entropy diagram of the cycle is shown in Fig. 1, where Q_{12} and Q_{34} are the amounts of heat exchange between the working substance and the heat reservoirs at temperature T_H and T_L during the two isothermal processes, Q_{23} and Q_{41} are the amounts of heat exchange between the working substance and the regenerator during the two isochoric processes, and V_H and V_L are, respectively, the maximum and minimum volumes and A_H and A_L are, respectively, the maximum and minimum surface areas of the gas system. All heats Q_{12} , Q_{34} , Q_{23} and Q_{41} are positive. In order to improve the performance of the cycle, a regenerator is often used in two isochoric processes. By using Eqs. (5) and (6), the amounts of heat exchange in the various processes of the micro/nano scaled Stirling cycle may be expressed as

$$Q_{12} = Nk_b T_H \left[\ln \frac{V_H}{V_L} + \left(\frac{A_L}{V_L} - \frac{A_H}{V_H} \right) \frac{1}{8C^{1/3} \sqrt{T_H}} \right], \quad (7)$$

$$Q_{34} = Nk_b T_L \left[\ln \frac{V_H}{V_L} + \left(\frac{A_L}{V_L} - \frac{A_H}{V_H} \right) \frac{1}{8C^{1/3} \sqrt{T_L}} \right], \quad (8)$$

$$Q_{23} = \int_1^2 C_V(T, A_H, V_H) dT = \frac{3}{2} Nk_b (T_H - T_L) + \frac{Nk_b}{8C^{1/3}} \frac{A_L}{V_L} (\sqrt{T_H} - \sqrt{T_L}) \quad (9)$$

and

$$Q_{41} = \int_3^4 C_V(T, A_L, V_L) dT = \frac{3}{2} Nk_b (T_H - T_L) + \frac{Nk_b}{8C^{1/3}} \frac{A_H}{V_H} (\sqrt{T_H} - \sqrt{T_L}), \quad (10)$$

respectively. From Eqs. (7)–(10), we can obtain the work input per cycle as

$$\begin{aligned} W &= Q_{12} - Q_{34} + Q_{23} - Q_{41} \\ &= Nk_b T_H \left[(1 - \tau) \ln r_V + \frac{L_c(T_H)}{2\sqrt{\pi}} \frac{A_L}{V_L} \left(1 - \frac{r_A}{r_V} \right) (1 - \sqrt{\tau}) \right], \end{aligned} \quad (11)$$

where $\tau = T_L/T_H$, $r_V = V_H/V_L$ and $r_A = A_H/A_L$. Because the net input work must be positive, we have $r_A < r_V [1 + 2\sqrt{\pi} V_L (1 + \sqrt{\tau}) \ln r_V / L_c(T_H) A_L]$.

Since the influence of surface area the amount of heat Q_{23} flowing into the regenerator in one isochoric process is different from the amount of heat Q_{41} flowing from the regenerator in the other isochoric process. Furthermore, by using the Eqs. (9)

and (10) the regenerative losses ΔQ_T is given by

$$\Delta Q_T = Q_{23} - Q_{41} = Nk_b T_H \frac{L_c(T_H)}{4\sqrt{\pi}} \frac{A_L}{V_L} \left(1 - \frac{r_A}{r_V}\right) (1 - \sqrt{\tau}). \quad (12)$$

From Eq. (12) we can observe that the value ΔQ_T is dependent of the volume ratio and the surface area ratio of two isochoric processes and can be positive, negative or zero according to the other parameters.

3.1. Case of $\Delta Q_T \geq 0$

In the case, according to the Eq. (12), $\Delta Q_T \geq 0$ means that the range of surface area ratio is $r_A \leq r_V$. However, considering the definition of r_A , then the range of surface area ratio is only $1 < r_A \leq r_V$. Further, the amount of heat exchange Q_{23} flowing into the regenerator in one isochoric process is larger than or equals that of heat exchange Q_{41} flowing from the regenerator in the other isochoric process. The redundant heat in the regenerator per cycle must be released to the cold reservoir in a timely manner because the present Stirling cycle is only operated between the two heat reservoirs T_H and T_L . If not, the temperature of the regenerator would be increased such that the regenerator would not operate normally between the T_H and T_L . Thus, the refrigeration heat R_{QB} absorbed from the cold reservoir is obtained as

$$R_{QB} = Q_{34} - \Delta Q = Nk_b T_H \left[\tau \ln r_V + \frac{A_L L_c(T_H)}{4V_L \sqrt{\pi}} \left(1 - \frac{r_A}{r_V}\right) (2\sqrt{\tau} - 1) \right]. \quad (13)$$

For an ideal gas confined in a macro scaled domain, the second term in Eq. (13) equals zero because $L_c(T) \ll V/A$ and the quantum boundary effect can be neglected. Therefore, the refrigeration heat of classical cycle can be written as

$$R_C = Nk_b T_H \tau \ln r_V. \quad (14)$$

However, in micro/nano scale the thermal wavelength $L_c(T)$ of gas atoms may be comparable with the size V/A of the domain and the quantum boundary effect cannot be neglected. Especially, the quantum boundary effect of gas particles is more important as the size of the finite domain is smaller. Therefore, in these cases the second term must be considered. From Eqs. (11) and (13), the coefficient of performance of cycle in the case can be expressed as

$$\varepsilon_{QB} = \frac{R_{QB}}{W} = \frac{\tau + \frac{A_L L_c(T_H)}{4V_L \sqrt{\pi} \ln r_V} \left(1 - \frac{r_A}{r_V}\right) (2\sqrt{\tau} - 1)}{1 - \tau + \frac{L_c(T_H)}{2\sqrt{\pi} \ln r_V} \frac{A_L}{V_L} \left(1 - \frac{r_A}{r_V}\right) (1 - \sqrt{\tau})}. \quad (15)$$

Furthermore, when $r_A = r_V$, the COP of the cycle equals the ideal Carnot COP

$$\varepsilon_{QB} = \tau / (1 - \tau) = \varepsilon_C. \quad (16)$$

From Eqs. (13) and (15) we can find that the refrigeration heat and COP of micro/nano scaled cycle are not only dependent of the volume ratio of two isochoric processes but also the surface area ratio of two isochoric processes when $1 < r_A < r_V$.

3.2. Case of $\Delta Q_T < 0$

In the case, similarly, according to the Eq. (12), $\Delta Q_T < 0$ means that the range of surface area ratio is $r_V < r_A$. However, considering the condition $W > 0$, then the range of surface area ratio is only $r_V < r_A < r_V [1 + 2\sqrt{\pi} V_L (1 + \sqrt{\tau}) \ln r_V / L_c(T_H) A_L]$. Especially, in the case the amount of heat exchange Q_{23} flowing into the regenerator in one isochoric process is smaller than that of heat exchange Q_{41} flowing from the regenerator in the other isochoric process. Moreover, the inadequate heat in the regenerator per cycle must be compensated from the hot reservoir in a timely manner because the present Stirling cycle is only operated between the two heat reservoirs T_H and T_L , while the refrigeration heat R_{QB} is unvarying when $\Delta Q_T < 0$. If not, the temperature of the regenerator would be changed such that the regenerator would not operate normally between the T_H and T_L . Thus, the refrigeration heat R_{QB} absorbed from the cold reservoir is obtained as

$$R_{QB} = Q_{34} = Nk_b T_H \left[\tau \ln r_V + \frac{A_L \sqrt{\tau} L_c(T_H)}{4V_L \sqrt{\pi}} \left(1 - \frac{r_A}{r_V}\right) \right]. \quad (17)$$

From Eqs. (11) and (17), the COP of the cycle can be expressed as

$$\varepsilon_{QB} = \frac{R_{QB}}{W} = \frac{\tau + \frac{A_L \sqrt{\tau} L_c(T_H)}{4V_L \sqrt{\pi} \ln r_V} \left(1 - \frac{r_A}{r_V}\right)}{1 - \tau + \frac{L_c(T_H)}{2\sqrt{\pi} \ln r_V} \frac{A_L}{V_L} \left(1 - \frac{r_A}{r_V}\right) (1 - \sqrt{\tau})}. \quad (18)$$

In the case, according to the Eqs. (17) and (18), the refrigeration heat and COP of the cycle also depend strongly on the surface area ratio of two isochoric processes besides that the volume ratio of cycle and other parameters. Eqs. (13), (15), (17) and (18) are the main results of this paper. Further, by using these equations, the performance characteristics of the micro/nano scaled Stirling refrigeration cycle may be analyzed in detail.

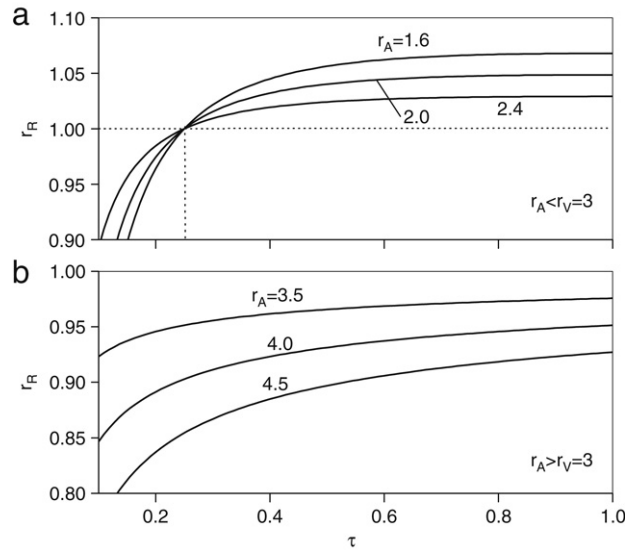


Fig. 2. The refrigeration heat ratio r_R versus the temperature ratio of cycle curves for the possible surface area ratios at given $r_V = 3$ and $\lambda = 0.16$; (a) $1 < r_A < r_V$, (b) $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L(1 + \sqrt{\tau}) \ln r_V/L_c(T_H)A_L]$.

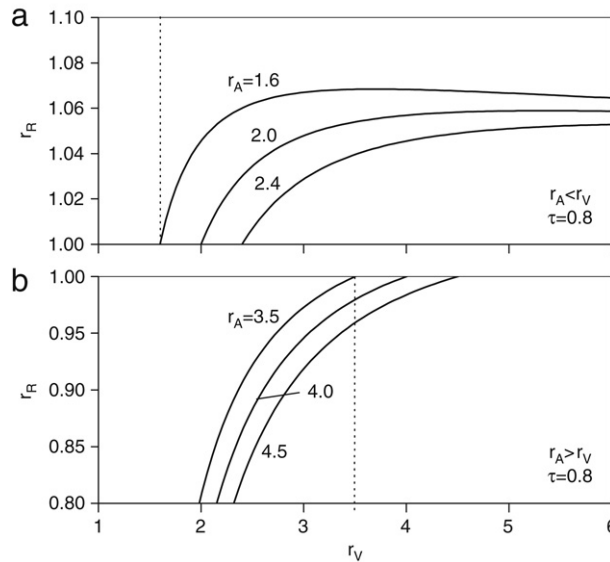


Fig. 3. The refrigeration heat ratio r_R versus the volume ratio of cycle curves for the possible surface area ratios at given $\tau = 0.8$ and $\lambda = 0.16$; (a) $1 < r_A < r_V$, (b) $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L(1 + \sqrt{\tau}) \ln r_V/L_c(T_H)A_L]$.

In following paper, for convenience of analysis, one may introduce the refrigeration heat ratio $r_R = R_{QB}/R_C$, the COP ratio $r_\varepsilon = \varepsilon_{QB}/\varepsilon_C$ and the dimensionless thermal wavelength $\lambda = \frac{A_L L_c(T_H)}{4\sqrt{\pi}V_L}$. For a given $\lambda = 0.16$ and some different surface area ratios in the two ranges of $1 < r_A < r_V$ and $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L(1 + \sqrt{\tau}) \ln r_V/L_c(T_H)A_L]$ in micro/nano scaled system, the refrigeration heat ratio r_R versus the temperature ratio (at given $r_V = 3$) and volume ratio (at given $\tau = 0.8$) of cycle curves can be plotted, respectively, as shown in Figs. 2 and 3. It is clearly seen from Fig. 2 (a) in the case that the refrigeration heat ratio r_R increases as τ increases at given other parameters. Especially, the value of r_R is larger than one and as a decreasing function of the surface area ratio r_A as $\tau > 0.25$ but smaller than one and as an increasing function of the surface area ratio r_A as $\tau < 0.25$. Moreover, for a given value of $1 < r_A < r_V$ and $\tau = 0.8$, one can observe from Fig. 3(a) that r_R is not a monotonous function of r_V and exists a maximum value at the special value of r_V .

We can find from Fig. 2(b) and 3(b) that the refrigeration heat ratio r_R is less than one in the case of $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L(1 + \sqrt{\tau}) \ln r_V/L_c(T_H)A_L]$ and increases as τ and r_V increase at given other parameters. Further, in the

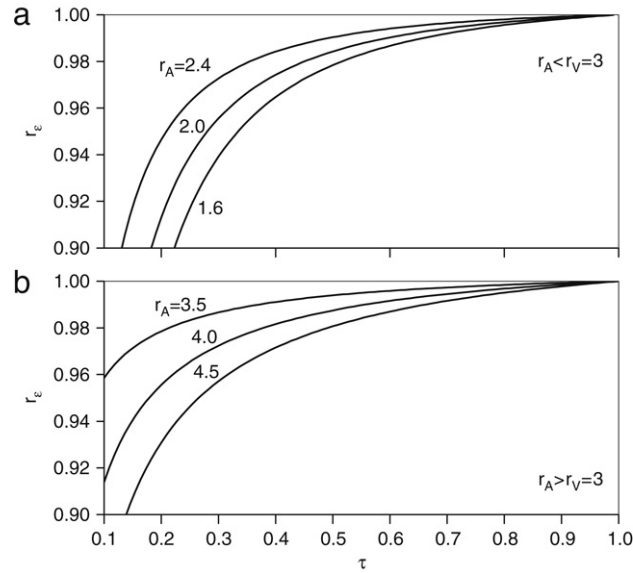


Fig. 4. The coefficient of performance ratio r_e versus the temperature ratio of cycle curves for the possible surface area ratios at given $r_V = 3$ and $\lambda = 0.16$; (a) $1 < r_A < r_V$, (b) $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L (1 + \sqrt{\tau}) \ln r_V/L_c (T_H) A_L]$.

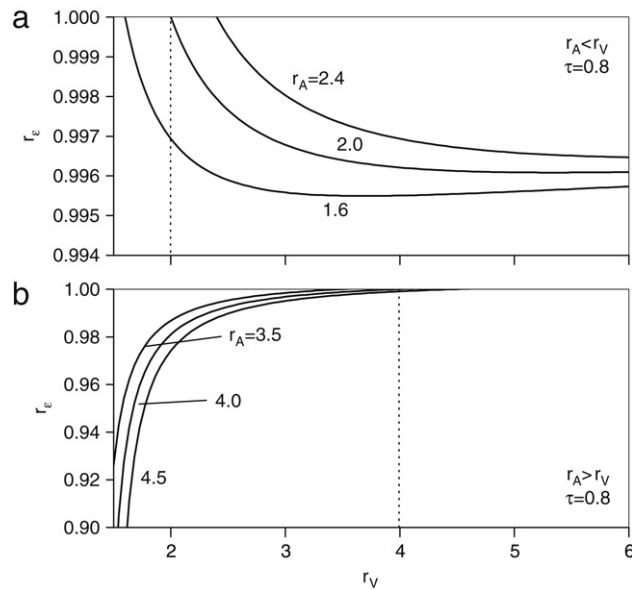


Fig. 5. The coefficient of performance ratio r_e versus the volume ratio of cycle curves for the possible surface area ratios at given $\tau = 0.8$ and $\lambda = 0.16$; (a) $1 < r_A < r_V$, (b) $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L (1 + \sqrt{\tau}) \ln r_V/L_c (T_H) A_L]$.

case the refrigeration heat always decreases as the surface area ratio r_A increases. Figs. 4 and 5 give the curves of the COP ratio r_e of cycle varying with the temperature ratio (at given $r_V = 3$) and volume ratio (at given $\tau = 0.8$) of cycle for different surface area ratio in the range of $1 < r_A < r_V [1 + 2\sqrt{\pi}V_L (1 + \sqrt{\tau}) \ln r_V/L_c (T_H) A_L]$. The COP ratio r_e of cycle is a monotonous increasing function of the temperature ratio τ and the surface area ratio r_A but not a monotonous function of r_V , and exists a minimum value at the special value of r_V at a given range $1 < r_A < r_V$. For the case of $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L (1 + \sqrt{\tau}) \ln r_V/L_c (T_H) A_L]$, the COP ratio r_e increases as τ or r_V increases but decreases as r_A increases at given other parameters. In particular, the COP ratios r_e of cycle in two cases are always less than one and the COP of the cycle may be improved when the volume ratio r_V approaches at the value of the surface area ratio r_A then the COP of cycle equals the ideal Carnot COP ε_C . It is noticed that the results obtained here based on the boundary effect of gas particles come about when the de Broglie's wavelength is comparable to the dimensions of the system.

4. Conclusions

In the paper, by considering the quantum boundary effect of gas particles, which appears if the system is confined in a finite domain, the performance characteristics of a Stirling refrigeration cycle in micro/nano scale are reanalyzed. It is found that a micro/nano scaled Stirling refrigeration cycle working with an ideal gas does not possess the condition of perfect regeneration due to the existence of quantum boundary effect in a finite domain. Furthermore, considering the boundary effects, the refrigeration heat and COP of the micro/nano scaled Stirling refrigeration cycle are derived. They are the functions of the temperature ratio τ of the two heat reservoirs, the volume ratio r_V and surface area ratio r_A of two isochoric processes and so on. In particular, the influence of the boundary of domain i.e. surface area on the performance of a micro/nano scaled Stirling refrigeration cycle is analyzed by introducing r_R and r_ε . It is found that the refrigeration heat of the cycle may be large than the refrigeration heat of the macro scaled Stirling refrigeration cycle as $1 < r_A < r_V$. That is, more heat is absorbed per cycle from cold reservoir. In spite of the fact, the COP of the cycle cannot exceed the COP of the macro scaled Stirling refrigeration cycle ε_C , since the corresponding work input per cycle must increase. However, as mentioned in Ref. [3], the number of cycles for refrigerator is an important performance parameter. In a real refrigerator, lost work also increases and COP decreases with an increasing number of cycles due to the friction losses. Consequently, it is understood that in the case of $1 < r_A < r_V$ the quantum boundary effect can be advantageous for a micro/nano scaled Stirling refrigeration cycle since the same refrigeration heat is obtained with a lower number of cycles. In the opposite direction i.e. $r_V < r_A < r_V [1 + 2\sqrt{\pi}V_L(1 + \sqrt{\tau}) \ln r_V/L_c(T_H)A_L]$, the refrigeration heat of the cycle is always smaller than the refrigeration heat of the macro scaled Stirling refrigeration cycle. Further, in the case the corresponding work input per cycle decreases and the COP of the cycle is also always smaller than the COP of the macro scaled cycle ε_C . Thus, in the case, the quantum boundary effect is always disadvantageous because it causes not only an increment in the number of cycles for given refrigeration heat but also the decrement of the COP of the cycle. Of course, for the present micro/nano scaled cycle, the COP of the cycle may be as large as possible and approach the ideal Carnot COP ε_C , by selecting properly the surface area ratio r_A of cycle approaching the volume ratio r_V of cycle. The results of the paper are very important and useful for designing of the micro/nano scaled Stirling refrigeration cycle. In addition, when selecting the values of surface area ratio r_A of cycle deviating from the value of volume ratio r_V of the cycle, the COP of cycle decreases largely. Thus this case leads to confirming experimentally the quantum boundary effect in micro/nano scaled devices.

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References

- [1] A. Sisman, H. Saygin, Appl. Energy 68 (2001) 367.
- [2] H. Saygin, A. Sisman, J. Appl. Phys. 90 (2001) 3086.
- [3] H. Saygin, A. Sisman, Appl. Energy 69 (2001) 77.
- [4] J. He, J. Chen, B. Hua, Appl. Energy 72 (2002) 541.
- [5] B. Lin, Y. Zhang, J. Chen, Appl. Energy 83 (2006) 513.
- [6] A. Sisman, H. Saygin, Phys. Scr. 63 (2001) 263.
- [7] E. Geva, R. Kosloff, J. Chem. Phys. 96 (1992) 3054.
- [8] E. Geva, R. Kosloff, Phys. Rev. E 49 (1994) 3903.
- [9] T. Feldmann, E. Geva, R. Kosloff, P. Salamon, Am. J. Phys. 64 (1996) 485.
- [10] F. Wu, L. Chen, F. Sun, C. Wu, P. Hua, Energy Convers. Manage. 39 (1998) 1161.
- [11] F. Wu, L. Chen, F. Sun, C. Wu, Int. J. Eng. Sci. 38 (2000) 239.
- [12] J. He, J. Chen, B. Hua, Phys. Rev. E 65 (2002) 036145.
- [13] M. Reith, Nano-Engineering in Science and Technology, World Scientific, New Jersey, 2003.
- [14] G. Timp, Nanotechnology, Springer, New York, 1999.
- [15] J.M. Saleh, Fluid Flow Handbook, McGraw-Hill, New York, 2002.
- [16] A. Sisman, I. Muller, Phys. Lett. A 320 (2004) 360.
- [17] A. Sisman, J. Phys. A: Math. Gen. 37 (2004) 11353.
- [18] A. Sisman, Z.F. Ozturk, C. Firat, Phys. Lett. A 362 (2007) 16.
- [19] H. Pang, W.S. Dai, M. Xie, J. Phys. A: Math. Gen. 39 (2006) 2563.
- [20] R.M. Ziff, G.E. Uhlenbeck, M. Kac, Phys. Rep. C 32 (1977) 169.
- [21] W.S. Dai, M. Xie, Phys. Lett. A 311 (2003) 340.
- [22] W. Nie, J. He, Phys. Lett. A 372 (2008) 1168.
- [23] W. Nie, J. He, X. He, J. Appl. Phys. 103 (2008) 114909.